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COMPARISON OF OPEN AND CLOSED BRAYTON ENGINES FOR RECUPERATED A--ETC(U)
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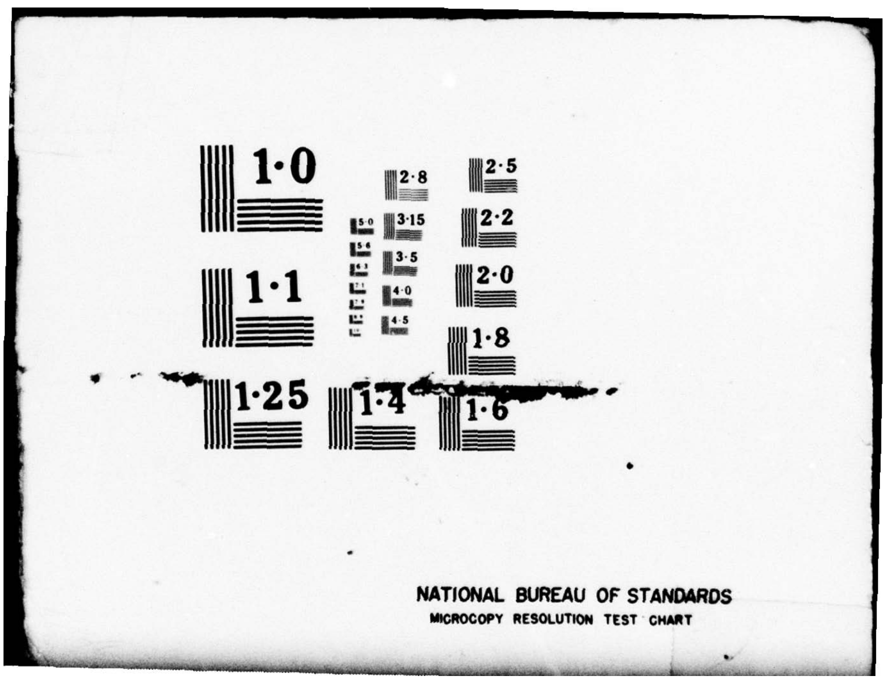
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6 COMPARISON OF OPEN AND CLOSED BRAYTON ENGINES FOR RECUPERATED AND SIMPLE CYCLES

by

10 John S./Wohlgemuth

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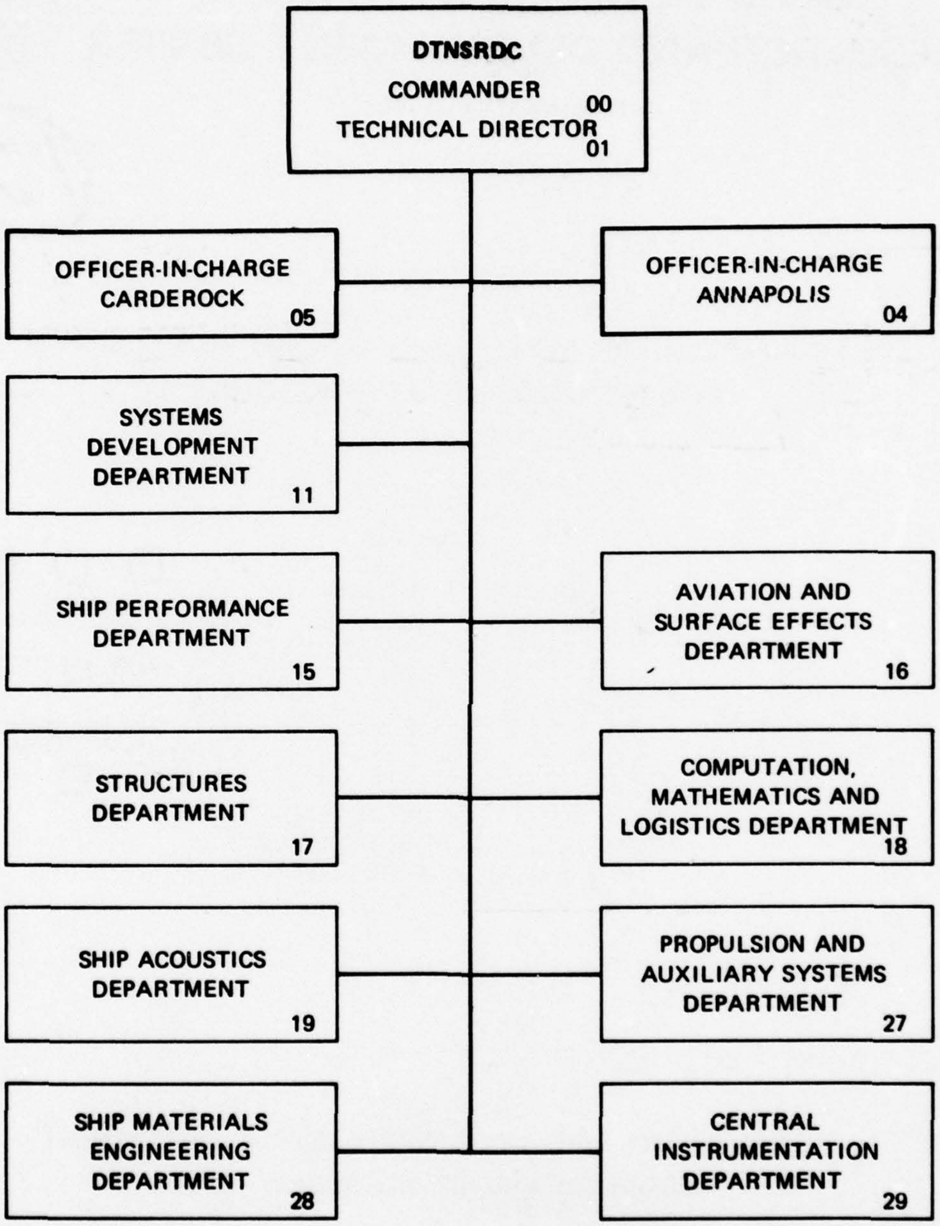

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To quantitatively compare open and closed Brayton systems for their possible recuperated and simple cycle versions, a general performance analysis is developed. The results of this analysis are summarized in open and closed cycle design point performance maps. The maps present the performance output (such as cycle efficiency) → next page

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20. ABSTRACT (cont)

over ranges of design values for the important cycle parameters of recuperation, gas turbine inlet temperature, and pressure ratio. For possible open and closed Brayton systems, cycle efficiency increases with either increasing values of recuperation or turbine inlet temperature, or both. To quantify these performance effects, the open and closed cycles are optimized with respect to pressure ratio. The efficiency improvement of recuperated open and closed Brayton cycles over simple cycles is about +8 points (for typical values of effectiveness of 0.90). This increase in efficiency for recuperated cycles holds over a reasonable range of turbine inlet temperatures of 1400°F (760°C) to 2200°F (1200°C). Since higher values of cycle efficiency can be achieved with recuperated cycles compared to simple cycles at constant turbine inlet temperature; the corollary also holds that, for constant values of efficiency, turbine inlet temperature will be lower for recuperated cycles compared to simple cycles. The reduction in turbine inlet temperatures at constant efficiency for recuperated over simple cycles is about 400°F (220°C) over a reasonable range of cycle efficiencies of 30% to 45%. Up to a recuperator effectiveness of 0.85, the open Brayton cycles have lower required turbine inlet temperatures to achieve the same cycle efficiencies as the closed Brayton cycles. For highly recuperated cycles (effectivenesses of 0.90 to 0.95), the open and closed Brayton cycles have competitive turbine inlet temperatures for competitive cycle efficiencies.

(Author)

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LIST OF ABBREVIATIONS

Btu	- British thermal unit
°C	- degree Celsius
CIT	- compressor inlet temperature
cm^2	- square centimeters
DLPB	- pressure drop for combustor or burner heat exchanger
DLPK	- pressure drop for cooler
DLPHXH	- pressure drop for recuperator heat exchanger, high-pressure cold-side
DLPHXL	- pressure drop for recuperator heat exchanger, low-pressure hot-side
°F	- degree Fahrenheit
F	- fuel flow
FG	- cycle addition of fuel flow
hp	- horsepower
in.	- inch
in.^{2}	- square inch
J	- joule
°K	- degree Kelvin
kg	- kilogram
kW	- kilowatt
lb	- pound

lb_f	- pound force
m	- meter
m^2	- square meter
MB	- gas flow, bleed
MC	- gas flow, compressor
MCT	- gas flow, compressor turbine
MMIX	- gas flow, mixed turbine exit with bleed
MPT	- gas flow, power turbine
MT	- gas flow, turbine expansion
N	- newton
psia	- pounds per square inch absolute
°R	- degree Rankine
RC	- pressure ratio, compressor, = P_2/P_1
RCT	- pressure ratio, compressor turbine
RIT	- recuperator inlet temperature
RPT	- pressure ratio, power turbine
sec	- second
SFC	- specific fuel consumption
SPHP	- net output power
TIT	- turbine inlet temperature
WC	- compressor power absorbed
WCT	- compressor power supplied

LIST OF SYMBOLS

CPC - constant pressure specific heat through compressor, $\frac{\text{Btu}}{\text{lb-°R}}\left[\frac{\text{J}}{\text{kg-°K}}\right]$

CPT - constant pressure specific heat through turbine, $\frac{\text{Btu}}{\text{lb-°R}}\left[\frac{\text{J}}{\text{kg-°K}}\right]$

P - pressure, lbf/in.^{2} (N/cm^2)

s - entropy, Btu/lb-°R (J/kg-°K)

T - temperature, °R (°K)

$\Delta P/P]_r$ - recuperator pressure loss

γ_C - ratio of specific heats through compressor

γ_T - ratio of specific heats through turbine

ϵ_r - recuperator effectiveness

η_B - burner or external heater efficiency

η_C - compressor efficiency

η_{GG} - gas generator efficiency

η_T - turbine efficiency

η_{TH} - thermal cycle efficiency

Numerical Subscripts

1 - compressor inlet

2 - compressor exit

5 - heater inlet

6 - maximum cycle temperature and TIT

8 - compressor (absorbed) turbine exit and intermediate expansion state-point

9 - final turbine expansion state-point

9R - cooled turbine exhaust and recuperator inlet (for recuperated cycles

10 - first heat rejection state-point

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ABSTRACT

To quantitatively compare open and closed Brayton systems for their possible recuperated and simple cycle versions, a general performance analysis is developed. The results of this analysis are summarized in open and closed cycle design point performance maps. The maps present the performance output (such as cycle efficiency) over ranges of design values for the important cycle parameters of recuperation, gas turbine inlet temperature, and pressure ratio. For possible open and closed Brayton systems, cycle efficiency increases with either increasing values of recuperation or turbine inlet temperature, or both. To quantify these performance effects, the open and closed cycles are optimized with respect to pressure ratio. The efficiency improvement of recuperated open and closed Brayton cycles over simple cycles is about +8 points (for typical values of effectiveness of 0.90). This increase in efficiency for recuperated cycles holds over a reasonable range of turbine inlet temperatures of 1400 degrees F (760 degrees C) to 2200 degrees F (1200 degrees C). Since higher values of cycle efficiency can be achieved with recuperated cycles compared to simple cycles at constant turbine inlet temperature; the corollary also holds that, for constant values of efficiency, turbine inlet temperature will be lower for recuperated cycles compared to simple cycles. The reduction in turbine inlet temperatures at constant efficiency for recuperated over simple cycles is about 400 degrees F (220 degrees C) over a reasonable range of cycle efficiencies of 30 percent to 45 percent. Up to a recuperator effectiveness of 0.85, the open Brayton cycles have lower required turbine inlet temperatures to achieve the same cycle efficiencies as the closed Brayton cycles. For highly recuperated cycles (effectiveness of 0.90 to 0.95), the open and closed Brayton cycles have competitive turbine inlet temperatures for competitive cycle efficiencies.

INTRODUCTION

The comparison of open and closed Brayton systems for their recuperated and simple cycle versions will be fundamentally based on their common thermodynamic cycle. The four processes of the general thermodynamic Brayton cycle common to both open and closed systems are in order: adiabatic (no heat transfer) compression of the gas, constant-pressure heat addition to the gas, adiabatic expansion of the gas, and finally constant-pressure heat rejection to return the gas to its original conditions. If the compression and expansion of the gas are accomplished by turbomachinery, then the Brayton cycle can be termed the gas turbine cycle also.

In a gas turbine cycle, the turbine exhaust is nearly always appreciably hotter than the gas exiting the compressor. Obviously the amount of heat input to the cycle can be reduced by the use of a heat exchanger in which the hot turbine exhaust gas gives up heat to the relatively cool gas from the compressor. Gas turbine arrangements that have such heat exchangers are called regenerated or recuperated, as opposed to the simple cycle versions that do not. What distinguishes open from closed systems is the manner in which heat addition to -- and heat rejection from -- the Brayton cycles are accomplished.
In the open Brayton cycle, heat addition takes place in a combustor by burning the fuel in the working air stream, with the resultant products of combustion mixing with and becoming part of the working gas. Open Brayton cycle heat rejection is accomplished by discharging the low-pressure exhaust gas into the system surroundings and recharging the open Brayton cycle with ambient air. The closed Brayton cycle gas flow is not allowed to mix with the system surroundings; therefore heat addition and heat rejection must

be accomplished through heat exchangers: a high-temperature heater and a low-temperature cooler, respectively. The manner in which the general Brayton cycle analysis defines the pertinent component parameters and models practical open and closed gas turbine systems for performance comparisons are detailed in the two Description sections found later in this report.

The development of advanced gas turbines will rely, at least partially, on improvements in their component designs. How great will be the performance improvements of advanced gas turbines is also dependent on the candidate systems developed. There are many options for Brayton cycle component improvements that the gas turbine industry may consider[1-3]. The objective of this study is to present a basis for comparison of open and closed Brayton cycle systems that will approximate the performance of possible advanced gas turbine candidates.

To determine where open and closed Brayton systems for possible recuperated and simple cycle versions are competitive, a general performance analysis is developed. This analysis utilizes a universal Brayton cycle (for open and closed, simple and recuperated systems) flow schematic with the flexibility to model most gas turbine configurations (from single to multiple stages) to determine the performance effects of the important component parameters over ranges of feasible design values. The approach used to summarize the numerous gas turbine possibilities was to generate a detailed performance map for each type of Brayton system (open and closed). The maps present the performance results

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[1]Superscripts refer to similarly numbered entries in the References at the end of the text.

over ranges of gas turbine pressure ratio (RC*), maximum turbine inlet temperature (TIT), and recuperator effectiveness (ε_r). These performance maps over ranges of RC, TIT and ε_r present possible choices for the important gas turbine component parameters along with their tradeoff effects as their design limits are approached (as detailed in the Approach and Results sections of this report). The ability of the analysis to output performance values such as thermal cycle efficiency (η_{TH}) and specific fuel consumption (SFC), over ranges of variable component parameters (such as RC, TIT, and ε_r), allows for selecting optimum values for design choices.[4-5] It is advantageous for one to select a system that uses the best component design values available for the material technology the particular application can utilize.[6-7] Furthermore, performance gains due to possible improvements in component technology [8-12] (e.g., higher values of ε_r or TIT) can be approximated by this study, as shown in the Conclusions section of this report).

DESCRIPTION OF UNIVERSAL BRAYTON CYCLE

GENERAL BRAYTON CYCLE

Figure 1 is the basic thermodynamic (T-s) diagram for the general Brayton cycle.[13,14] This general Brayton cycle illustrates the state-points and processes common to both open and closed gas turbines for their recuperated and simple cycle versions. All the processes are irreversible, meaning that actual component pressure losses and turbomachinery inefficiencies are considered. The total gas flow, MC, is compressed from its lowest temperature, T_1, to a higher temperature and pressure state-point 2. The ideal process

*Definitions of abbreviations appear on pages i and ii.

to achieve the pressure ratio ($RC=P_2/P_1$) is the reversible adiabatic or isentropic compression. The actual compressor power needed is greater than the ideal by the reciprocal of the compressor efficiency (η_c), where

$$\eta_C = \frac{\text{ideal compression power}}{\text{actual compressor power absorbed.}}$$

At the compressor exit, a small portion of the gas flow may be bled-off (MB) to supply turbine cooling, gas bearing, or other turbomachinery requirements, after which it returns to the cycle at the turbine exit, state-point 9.

The major portion of the gas flow follows a heat addition process from compressor exit to turbine inlet. During the initial part of the heat addition process, the gas temperature is raised from compressor exit temperature through intermediate temperatures which are still lower than the final turbine exhaust temperature, T_{9R}. Therefore, heat transfer is possible from the hot-side (turbine-exhaust-side) to the cold-side (compressor-exit-side) if an intermediate heat exchanger (recuperator) is included. Open and closed Brayton cycles with such intermediate heat exchangers are termed "recuperated cycles as opposed to the "simple cycle." Open and closed recuperated cycles have the performance advantage of utilizing heat from the turbine-exhaust-side that would otherwise have to be rejected. This heat transfer to the cycle heat addition process reclaims (or "regenerates") some of the cycle's waste heat, which in turn reduces the heat addition required externally to the cycle (from state-points 5 to 6). A reduction in external heat addition increases the cycle efficiency (η_{TH}). This recuperation effect can be quantified by the heat exchanger performance parameter, called effectiveness ϵ_r. Effectiveness is defined as the actual heat transferred divided by maximum heat

transfer possible between the temperature limits of the two flow streams.[14,15] Higher recuperated (effectiveness) open and closed Brayton cycles require less external heat addition than lower recuperated cycles and the simple Brayton cycle ($\varepsilon_r = 0$). Therefore, as ε_r increases, η_{TH} increases with all other performance parameters (T_6, ηc, RC, pressure losses, T_1, η_T) remaining fixed. Effectiveness is not the only cycle performance effect of the recuperator; the other main effect is the additional pressure loss (parameter $\Delta P/P]_r$) it adds to the open or closed Brayton cycle.

The cold-side of the recuperator has a pressure loss (DLPHXH) that is larger than the pressure loss associated with the portion of the heater heat exchanger it replaces in the closed Brayton cycle, or the pressure loss associated with the portion of the heater combustor it replaces in the open Brayton cycle. There is also pressure loss (DLPHXL) associated with the hot-side of the recuperator for both open and closed Brayton cycles. These additional pressure losses (DLPHXH and DLPHXL) have a negative effect on the cycle performance (η_{TH}) for any Brayton cycle, open or closed. Therefore, in performance comparisons for recuperated and simple cycles, considerations of the positive influence of effectiveness (ε_r) and the negative influence of additional pressure losses (DLPHXH and DLPHXL) must be made. The performance gains for increasingly higher values of effectiveness (ε_r's) should exceed the performance penalties of the associated higher recuperator pressure losses ($\Delta P/P]r$'s, where $\Delta P/P]_r$ = DLPHXH and DLPHXL) for both open and closed Brayton systems (although the performance tradeoffs for open and closed are different).

After internally transferring heat across a recuperator, if one is used, externally supplied heat addition (distinguished by a combustor in open cycles and a heater heat exchanger in closed cycles) raises the gas flow to its maximum cycle temperature, (T_6), which is also taken to be turbine inlet temperature (TIT). There is an associated pressure loss (DLPB) for this external heat addition. The pressure ratio now available for open or closed Brayton turbine expansion (P_6/P_9) has been reduced due to the pressure losses of the heat transfer components mentioned above. There is one additional pressure loss for the closed Brayton cycle cooler (DLPK) that must be accounted for in closed systems.

Part of the turbine expansion power is absorbed driving the compressor, which is shown in Figure 1, as process 6 to 8. The remaining turbine expansion (pressure ratio, P_8/P_9) produces the net open or closed Brayton cycle power. The turbine gas flow (MT) may change during the expansion process if blade cooling and/or staging is used. The actual turbine expansion power is lower than the ideal process (isentropic expansion) power by the turbine efficiency (η_{TH}), where

$$\eta_{TH} = \frac{\text{actual turbine power produced}}{\text{ideal expansion power}}.$$

Finally, heat rejection (state-points 10 to 1) returns the general (open and closed, recuperated or simple) Brayton cycle to its original state-point 1. Heat rejection in the open cycle is accomplished by discharging the low-pressure gas flow to the atmosphere. At state-point 1, the gas flow is recharged with ambient-temperature air from the atmosphere. Heat rejection in the closed cycle is accomplished by passing

the gas flow through a cooler until its temperature is returned to T_1.

FLOW SCHEMATIC WITH SYSTEM, COMPONENT AND ARRANGEMENT FLEXIBILITY

A flow schematic that models the Brayton cycle (Figure 1) for general mass flows is shown in Figure 2. This schematic is universal in that it can be used for closed or open Brayton systems by specifying cycle flow through a heat rejection cooler (denoted by the closed-loop component K), or as exiting the open cycle at an ambient pressure exhaust. Common nomenclature can be used for both closed and open systems, since mass balances have been satisfied for all possible flowpaths. Two of the more important mass balances are: (1) for the closed Brayton cycle, fuel-flow (FG) is not added to the gas-flow (this means for the closed cycle FG = 0); and (2) the gas-flow that is compressed must equal the gas-flow that expands through the turbine plus any by-pass flow (MC = MPT + MB) for either the closed or open Brayton systems. For the open cycle, the flow through the compressor turbine and power turbine stages is increased by the value of the fuel-flow (MCT + FG and MPT + FG, respectively). If there is any gas-flow addition between turbine stages, the turbine gas-flow must increase (MPT>MCT). The turbine stages that provide compressor power (WCT, where WCT = $WC/\eta GG$) and those that produce net output power (SPHP) may be physically separated into a compressor turbine component (denoted by CT) and a power turbine component (denoted by PT), or may be integrated into one turbine. If the open or closed Brayton system has an individual power turbine, it may be physically coupled to the compressor turbine (usually referred to as a single-shaft engine), or it may be decoupled from the other turbo-

machinery in a freepower turbine arrangement. This arrangement flexibility is shown in Figure 2 as dashed lines that represent a possible physically connected system. The flow schematic can model any of the above component arrangements by the selection of the proper set of numerical values for mass flows and component parameters.

As an example, for small-output-power Brayton engines, the turbine can be modeled as a single component with a single by-pass gas-flow, as in the case of the DTNSRDC Closed Brayton Laboratory Engine (CCPS 40-1).[16] When the true component design values were put into the flow model, the actual design performance of 31.0 percent was calculated.

The off-design efficiencies of specific gas turbines result from their off-design component parameters (such as η_c, η_T, and RC). There is a unique set of off-design component parameters for each case of part-power for a particular gas turbine. If the off-design component parameters are known, based on known values of part-power matchpoints, then their values can be put into the flow model and the off-design efficiencies for the particular gas turbine can be calculated. This approach was utilized by Bowen[17], where the component efficiencies were assumed as a function of power. However, what is usually known is the performance characteristics of the individual components of the particular gas turbine, and each part-power case must be solved by finding the matchpoint common to all the components. Such off-design performance predictions of specific gas turbine engines have been developed by others, e.g., the well-known NEPCOMP method[18] for simple cycle engines and the recent extension to closed Brayton cycle engines by Knauss.[19]

Advanced open and closed Brayton engines would be

expected to have multiple turbine stages with several by-pass gas-flow requirements. A general flow schematic for advanced open and closed systems would use separate turbine components for modeling turbine staging with the effect of by-pass flows included. Figure 2, with its complete set of gas-flows and components, can be used to generally model advanced open and closed Brayton systems. Potential performance improvements, for either open or closed systems, will be the results of improved component performances. As discussed above, major cycle performance improvements are possible for higher effectiveness recuperators if the heat exchanger temperatures can be tolerated. Likewise, performance improvements are possible for higher maximum temperature cycles (for both simple and recuperated). It may be simpler to increase the maximum turbine inlet temperature than to increase recuperator effectiveness, or vice versa, as the design limits of one or the other are approached. These component performance trade-offs change at different values of both ε_r and TIT in both open and closed Brayton systems, although again these parametric effects are dependent on the type of system (open or closed). By having a general flow schematic (Figure 2) that can model both open and closed Brayton systems over ranges of effectiveness and turbine inlet temperatures, performance trade-offs due to these component parameters can be examined for open and closed systems and comparisons between the types of Brayton systems can be made. The description of how the component design values were selected and analyzed by a general Brayton cycle program, and the approach used to generate performance trade-offs, follow in the next two sections.

DESCRIPTION OF BRAYTON CYCLE PROGRAM

INPUT FOR BRAYTON CYCLE ANALYSIS

To be able to compare open and closed Brayton systems for their current state of the art to possible advanced versions, it is necessary to choose values for the component parameters that can remain constant over ranges of pressure ratio (RC), maximum turbine inlet temperature (TIT), and recuperator effectiveness (ϵ_r). Values for these component efficiencies and pressure drops, as well as typical turbomachine flowratios, are given in Table 1. Since the burner efficiency (η_B) for the open cycle is synonymous to the combustor efficiency and does not include a heat exchanger efficiency (as does the closed cycle burner efficiency), η_B is three points higher for open cycle systems compared to closed cycle systems. The other component parameter that is different for the closed and open systems is the additional pressure drop DLPK the closed cycle must have for a cooler. It should be noted that since the effectiveness of state-of-the-art coolers is generally so high ($\approx$1.0), the compressor inlet temperature (CIT) for the closed cycle can be considered equivalent to that of the open cycle.

The only other fixed inputs for the open and closed Brayton cycle analysis are the working fluid properties. It is possible to use numerous gases in closed Brayton cycles, such as the inert gases argon, helium, etc. However, component performance (turbomachinery efficiencies and pressure losses) are thermodynamically and fluid mechanically dependent on the properties of each particular working gas. The ability to compare the performance (η_{TH}) of open and closed Brayton cycles as thermodynamically related to the amount of

recuperation (ϵ_r) or the maximum turbine inlet temperature (TIT), for ranges of their feasible design values, requires using the same working gas, air. Any differences in the performance (η_{TH}) of open and closed Brayton engines for their recuperated or simple cycles would, therefore, be related to differences in their physical component parameters (i.e., $\Delta P/P]_r$ for equivalent ϵ_r, as explained below) and not to differences in fluid properties due to different gases. It was not within the scope of this study to consider the physical effects on turbomachinery and recuperators that different closed cycle working fluids would have. Cycle temperature effects on air properties were considered as given in Table 2.

The remaining input parameter for the Brayton cycles, recuperator pressure drop ($\Delta P/P]_r$), must be varied over the range of recuperator effectiveness (ϵ_r) for both open and closed systems. A common design approach for heat exchangers is the NTU (number of transfer units) method.[20] A relationship to determine the NTU performance as a function of pressure drop has been derived by Aronson [21] for compact heat exchangers. This function can be described as

$$NTU = f_1(\Delta P/P]_r).$$

The performance can be related to the more classical performance of effectiveness when the heat exchanger geometry is considered,

$$\epsilon_r = f_2 \text{ (NTU, geometry)}.$$

This relationship has been extensively examined in texts such as Kays and London.[22] For a fixed geometry heat exchanger (recuperator), the curve shape, $\epsilon_r = f_2$ (NTU), has been derived. Using the approach as referenced above, a

relationship for $\epsilon_r = f_3\ (\Delta P/P]_r)$ can be found. Typical recuperator performance results are given in Figure 3.

Pressure losses ($\Delta P/P]r$) are lower in the closed recuperators (dashed line) than in the open recuperators (solid line). This can be explained by differences in their flow conditions at an equivalent open and closed recuperator performance rating (ϵ_r). Recuperator pressure loss ($\Delta P/P]_r$) has a direct relationship to its physical geometry and the velocity of the gas flow passing through it. Also, recuperator performance (ϵ_r) is related to its physical geometry and to its NTU rating as explained above (and in Kays and London).[22] NTU rating is further related to gas flow.[20] Therefore, for open and closed recuperators with equivalent physical geometry and performance (ϵ_r) and equivalent gas flow rates, any difference in pressure loss ($\Delta P/P]_r$) is due only to a difference in gas flow velocity. Gas flow velocity will be lower in the closed recuperator than in the open recuperator because of the advantage of pressurizing the closed Brayton cycle gas flow. Therefore, a closed cycle recuperator with equivalent physical geometry, performance (ϵ_r), and gas flow as an open cycle recuperator, will have a lower pressure loss ($\Delta P/P]_r$) than the open cycle recuperator due to the lower closed cycle flow velocity. The lower pressure loss ($\Delta P/P]_r$) for the closed recuperator can be seen in Figure 3. It is interesting to note that since Kays and London's recuperator performance (ϵ_r) curves [22] approach the value of 1.0 as the values of NTU approach infinity, and NTU is directly related to recuperator pressure loss ($\Delta P/P]_r$), the recuperator curves of Figure 3 would approach infinite values of $\Delta P/P]_r$ at $\epsilon_r = 1.0$. Open and closed recuperated Brayton cycles can tolerate only reasonable values of pressure loss ($\Delta P/P]_r$) before the recuperator pressure loss ($\Delta P/P]_r$) becomes disproportionately large compared to other component

pressure losses (i.e., DLPB). Therefore, ($\Delta P/P]_r$) for both open and closed recuperators was limited to 6 percent as a reasonable maximum value with regard to the other Brayton cycle pressure losses, as discussed above.

CALCULATION OF BRAYTON CYCLE PERFORMANCE

The overall energy conversion performance of a thermal cycle is usually quantified as the net output power (i.e., shaft) produced for the thermal input power required. The dimensionless conversion ratio of output divided by input is defined as thermal cycle efficiency (η_{TH}). η_{TH} is not dependent on the absolute magnitude of the output and the input, but is solely dependent on the degree to which the cycle converts the input into the output. To emphasize the fuel required to supply the thermal cycle input for a net output, the conversion ratio of fuel energy divided by output energy yields the performance quantity called specific fuel consumption (SFC). SFC is numerically dependent on the system of units used but, (as with η_{TH}), is independent of the magnitudes of the cycle output. η_{TH} and SFC are simply dimensionally different forms to express the same degree to which the cycle performs the overall energy conversion. To be independent of the absolute magnitudes of energies converted, both η_{TH} and SFC are independent of the rate with which the conversion occurs. The rate of energy conversion for a thermal cycle (open or closed) is directly dependent on the flow rate of its working fluid, which in turn is related to the absolute physical size of the components. Therefore, η_{TH} and SFC are independent of the cycle flow rate and its relationship to system physical size. Working fluid energies produce or absorb work in the turbomachinery. If the absolute flow rate is used in component performance calculations, then the rate of these

energy conversions, or cycle absolute powers, are calculated. If equivalent size components are used, then component performance may be calculated on a unit-flow-rate basis. Therefore, component powers may be calculated on an absolute or unit basis depending on a similar choice of flow rates; however, the overall cycle performance (η_{TH} or SFC) is independent of the flow rate choice, as mentioned above. Open or closed Brayton cycle performance is calculated by the computer subroutine listed in Appendix A. Component parametric values are put into the appropriate variables in the equations. The equations contain flowrate terms, so that absolute or per unit-flow powers (for WC, WCT and SPHP) can be calculated. Therefore, the subroutine can output either type of turbomachinery powers by inputting either absolute or unit flow rates (for MC, MCT, and MPT + FG, respectively). For example, net output power, SPHP, can be calculated in either absolute units (hp) for the parametric units in the equation, or as a specific value (hp/lb/sec) by inputting MPT and FG as either absolute flow rates (lb/sec) or as flow ratios of MC (lb/sec/lb/sec), respectively.

BRAYTON CYCLE PERFORMANCE OUTPUT

The performance of any Brayton cycle (open or closed) is usually quantified in terms of its temperature-and-pressure state-points and the output quantities defined above (i.e., η_{TH}, SFC, SPHP, etc.). Some of the open and closed Brayton cycle state-points are specified by the physical and material limits of their components (such as TIT and RIT), while other state-points are the results of the performance of the cycles. By using the general component parameters discussed above and utilizing the subroutine of Appendix A, open and closed Brayton cycle performance over ranges of pressure ratio (RC), maximum turbine inlet temperature (TIT), and recuperator effectiveness (ε_r) are calculated. These performance values

in the output listing are specific fuel consumption (SFC), thermal efficiency (η_{TH}), specific horsepower (SPHP) and fuel-air (working fluid) ratio. For each case of recuperator effectiveness (ε_r) with a specified turbine inlet temperature (TIT), there is a range of compressor pressure ratios that result in open or closed Brayton cycles with output power. Therefore, the open and closed Brayton analysis calculates the output performance over a range of practical pressure ratios for each case of recuperation and turbine inlet temperature. Figure 4 is an example of the output performance for a possible case of TIT and ε_r over a range of RC. In the example given (Figure 4), the value of RC = 3.00 would give the "optimum" or highest output thermal efficiency (η_{TH} = 0.422) for the case of TIT and ε_r given. By analyzing many cases of ε_r and TIT, the performance effects of the design parameters can be generated and the output compared for many possible Brayton cycles (open or closed). An approach to analyze open and closed Brayton cycle performance over possible ranges of the important component parameters is discussed in the next section.

APPROACH

To compare the performance of recuperated and simple Brayton cycles for open and closed systems, it is necessary to optimize the selection of certain component parameters, and to examine the effects on performance that variable design parameters have over reasonable ranges of their values. An approach to analyzing the many possible choices for either open or closed Brayton systems is to summarize the performance of numerous cycles in a design point performance map. Such maps present the performance output as the results of the more important component design parameters. As discussed above, open or closed Brayton cycle performance increases with increasing TIT and increasing values of recuperation (once

the pressure loss effects are exceeded). Therefore, performance maps (of open and closed Brayton cycles) for ranges of TIT and ε_r would present possible choices of these important component parameters and would compare their trade-offs when their design limits are approached.

For each case of recuperation analyzed, cycle thermal efficiency (η_{TH}) can be plotted as a function of TIT and RC. Figure 5 illustrates this cycle behavior approximately for simple Brayton cycles ($\varepsilon_r = 0$). It can be seen from Figure 5 that for each value of TIT there is a value for RC that gives a higher value of η_{TH} than any other RC and the same TIT. Hence, for every TIT there is an optimum choice for compressor pressure ratio (RC_{OPT}, shown as the solid line in Figure 5). The optimum value of pressure ratio (RC_{OPT}) for each combination of ε_r and TIT can be found in the computational output listing. The complete design point performance map can be generated from these optimum combinations of RC for TIT and ε_r values, as illustrated in Figure 6.

With a detailed performance map for each type of Brayton system (open or closed), comparisons of the effects of recuperation (ε_r) and maximum turbine inlet temperature (TIT) can be made. In addition, the open and closed Brayton systems can be compared at important parameter design choices to see if their performance is competitive. These detailed performance maps for open and closed Brayton cycles are presented in the next section.

RESULTS

As discussed in previous sections, performance maps summarize the results of each type of Brayton system (open and closed). Figures 7 and 8 present the performance maps of open and closed Brayton cycles, respectively. These figures plot

the resultant Brayton cycle efficiency (η_{TH}) over ranges of the following important component design-point parameters: maximum turbine inlet temperature (TIT), recuperator effectiveness (ε_r), and optimum compressor pressure ratio (RC_{OPT}).

As can be seen in Figures 7 and 8, the performance (η_{TH}) gains for increasingly higher values of effectiveness (due to the reduction in heat input to the open or closed Brayton cycle) do exceed the performance penalties of their associated pressure losses. For the open and closed Brayton systems (Figures 7 and 8, respectively), this performance tradeoff occurs at an effectiveness (ε_r) of approximately 0.60, with the performance advantage of recuperation becoming increasingly greater over the effectiveness range of $0.60 < \varepsilon_r < 0.95$. For current technology the performance improvement of recuperation in the open or closed Brayton cycle over the simple cycle is about + 8 points in η_{TH} (for $\varepsilon r = 0.90$). This comparison holds over a reasonable range of turbine inlet temperature of 1400 degrees F (760 degrees C)< TIT<2200 degrees F (1205 degrees C). The performance pay-off for increasing the amount of recuperation above $\varepsilon_r = 0.90$ is about 1/2 point in thermal cycle efficiency for each one point in effectiveness. This performance effect can be expressed as the recuperation influence factor

$$\frac{\eta_{TH}}{\varepsilon_r} \approx \frac{0.5 \text{ point}}{1 \text{ point}}$$

(for the ranges of $0.85 < \varepsilon_r < 0.95$ and 1400 degrees F (760 degrees C) < TIT < 2200 degrees F (1205 degrees C) in Figures 7 and 8) These results show quantitatively that the open or closed Brayton cycle performance (η_{TH}) effect of recuperation increases rapidly at high values of effectiveness (as does recuperator size). (It took a 90 point increase in ε_r (from 0.00 to 0.90) to get an 8-point gain in η_{TH}, but then only a 10-point increase in ε_r (from 0.85 to 0.95) to get a 5-point gain in η_{TH}).

Since higher values of η_{TH} can be achieved with higher values of ε_r at constant TIT's; the corollary also holds that for constant values of η_{TH}, lower values of TIT can be used with higher values of ε_r. An example of this, in Figure 7, is that for a constant value of η_{TH} = 35 percent, the required turbine inlet temperature with $\varepsilon_r = 0.90$ is TIT = 1325 degrees F (720 degrees C) which is 400 degrees F (220 degrees C) lower than the required turbine inlet temperature with $\varepsilon_r = 0.00$ of TIT = 1725 degrees F (940 degrees C). This reduction in TIT (for recuperated compared to the simple cycle) of approximately 400 degrees F (220 degrees C) holds for design cases of thermal cycle efficiency over the range of 30 percent $\leq \eta_{TH} \leq$ 45 percent for both open and closed systems.

The performance trade-offs when comparing recuperated to simple Brayton systems are, therefore, considerations of possible η_{TH} improvements if TIT can be maintained constant (over the range of ε_r), or the possible reductions in required TIT to maintain a constant η_{TH} in recuperated open or closed Brayton cycles.

When selecting the component parameters for a gas turbine engine, the open or closed Brayton cycle conditions must be compared to the limiting material capabilities. Material technology dictates not only the maximum turbine inlet temperature (TIT), but also the maximum recuperator inlet temperature (RIT) if the open or closed Brayton cycle is recuperated. Therefore, values for RIT are given in the performance maps (Figures 7 and 8) so that the material technology needed can be determined. For high values of TIT and ε_r, recuperator inlet temperatures range from 1200 degrees F (650 degrees C) to 1600 degrees F (870 degrees C), see Figures 7 and 8. McDonald and others [23-26] have surveyed recuperator technology over possible gas turbine performance ranges. It is suggested

that stainless steel recuperators can be used in advanced gas turbine applications for RIT's of 1200 degrees F (650 degrees C) to 1400 degrees F (760 degrees C), after which superalloy recuperators could be used to a maximum RIT of approximately 1800 degrees F (980 degrees C). Usually, marine gas turbine materials are limited to 1600 degrees F (870 degrees C) for uncooled parts [27] to avoid sulfidation corrosion. Therefore, in marine applications the maximum RIT should be limited to 1600 degrees F (870 degrees C).

As discussed above, with a detailed performance map for both types of Brayton systems (open and closed, Figures 7 and 8, respectively) possible candidates, specified by the design performance (η_{TH}) required, can be compared for competitiveness. The performances of open and closed systems will differ due to differences in their component parameters. Open Brayton systems have two performance advantages that closed Brayton systems do not have. As discussed above, these advantages are a higher heater efficiency (η_B) and fuel flow addition to the working fluid for the open cycle (see Figure 2). Closed Brayton recuperated systems have the performance advantage of lower pressure losses ($\Delta P/P]_r$) in the recuperator. The Figures 7 and 8 performance maps show where the advantages of one type of system are greater than the advantages of the other. The required TIT and ε_r values of different systems can be compared to determine if they are competitive. As an example, to obtain a performance of η_{TH} = 40 percent, open and closed candidate systems can be found in Figures 7 and 8. Up to an effectiveness (ε_r) of 0.85 the open Brayton systems have lower required TIT's than closed systems (e.g., 1650 degrees F (900 degrees C) compared to 1750 degrees F (955 degrees C) at ε_r = 0.85). For highly recuperated cycles, the open and closed systems have competitive TIT's of about 1620 degrees F (880 degrees C) at ε_r = 0.90 and 1480 degrees F (805 degrees C) at ε_r = 0.95. Other design efficiencies (η_{TH}'s) can be

compared similarly by Figures 7 and 8, for whatever performance the particular application requires.

CONCLUSIONS

It is advantageous to select an open or closed Brayton system that uses a combination of performance improvements and satisfactory component temperature and size requirements for the particular application. Figures 7 and 8 approximate the design performances of possible open and closed Brayton systems for the component technology that one may want to consider.

Furthermore, the performance gains due to possible improvements in component technology (e.g. higher values of TIT or ε_r) can be approximated. By examining the performance maps (Figures 7 and 8) for the particular design constraints, the performance pay-offs of possible candidate Brayton systems (open and closed) can rapidly be compared. These performance improvements can be quantified as approximately an increase of +8 points in η_{TH} or as a reduction of 400 degrees F (220 degrees C) in required TIT for recuperated open or closed Brayton cycles compared to simple cycles, or as other design trade-offs as discussed above. Up to an effectiveness (ε_r) of 0.85, the open Brayton systems have lower required turbine inlet temperatures (TIT's) to achieve the same cycle efficiencies (η_{TH}'s) as the closed Brayton systems. For highly recuperated cycles ($0.90 < \varepsilon_r < 0.95$), the open and closed systems have competitive TIT's for competitive η_{TH}'s.

In addition to being able to make performance comparisons of general open or closed Brayton cycle systems, the analysis developed can model any unique gas turbine

candidate whose design values are defined in typical component parameters. The ability of the analysis to calculate open or closed Brayton cycle performance values over selected ranges of variable design point parameters (such as RC, ε_r, and TIT) allows for the optimization of these design point parameters.

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Table 1 Component Input for Open and Closed Brayton Cycle Analysis

Parameter	Notation	Value	
		Open	Closed
Compressor efficiency	η_C	0.85	0.85
Turbine efficiency	η_T	0.90	0.90
Gas generator efficiency	η_{GG}	0.99	0.99
Burner efficiency	η_B	0.98	0.95
Pressure drop for burner	DLPB	0.03	0.03
Pressure drop for cooler	DLPK	-	0.001
Bleed gas flow	MB	0.01	0.01
Compressor turbine gas flow	MCT	0.97	0.97
Power turbine gas flow	MPT	0.99	0.99
Compressor inlet temperature	CIT	70°F (21°C)	70°F (21°C)

Table 2. Cycle Temperature Effects on Air Properties

Parameter	Notation	Value
Constant-pressure specific heat through compressor	CPC	$0.240 \frac{Btu}{lb-°R}$ $(1000 \frac{J}{kg-°K})$
Ratio of specific heats through compressor	γ_C	1.400
Constant-pressure specific heat through turbine	CPT	$0.274 \frac{Btu}{lb-°R}$ $(1050 \frac{J}{kg-°K})$
Ratio of specific heats through turbine	γ_T	1.330

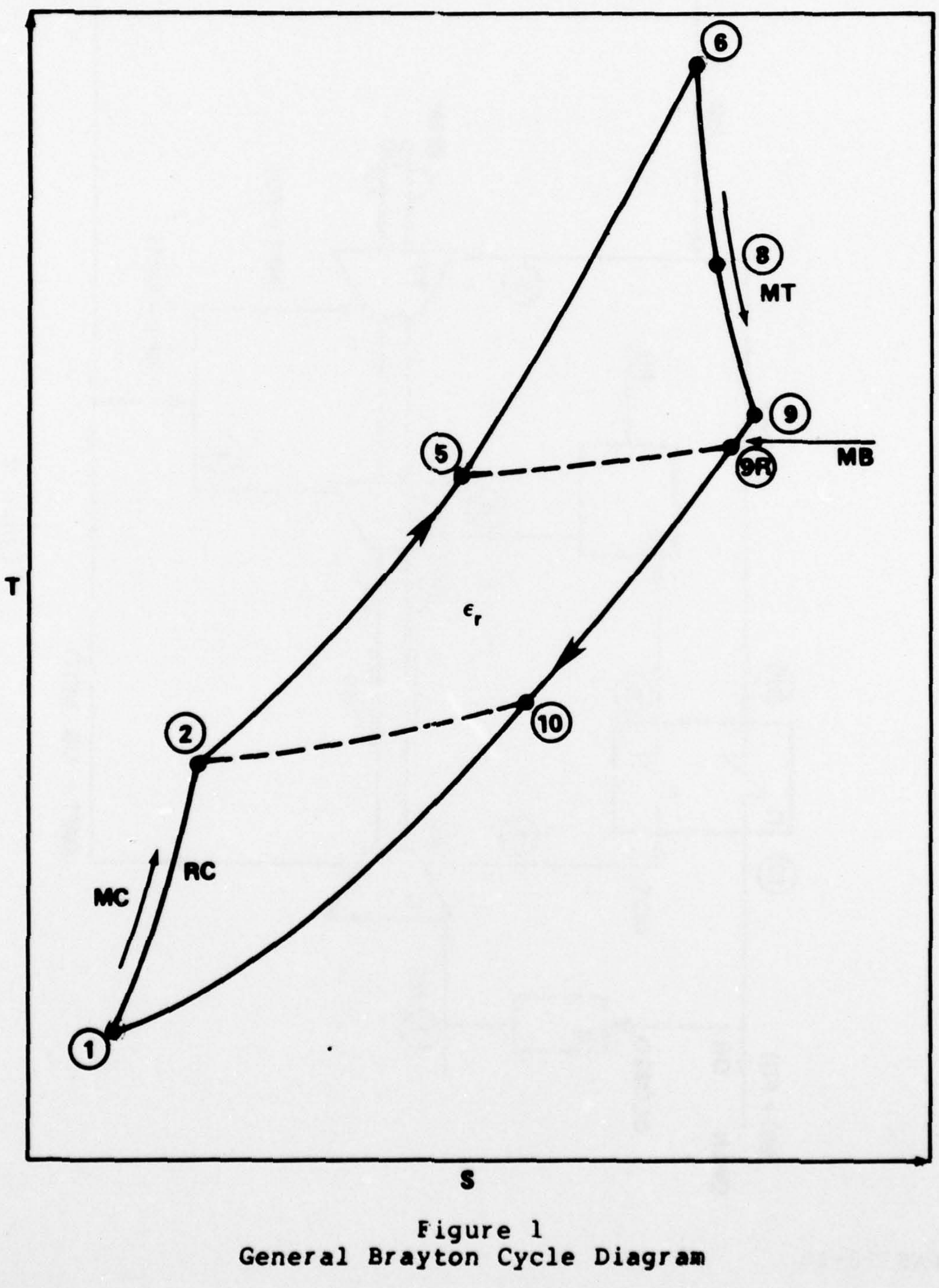

Figure 1
General Brayton Cycle Diagram

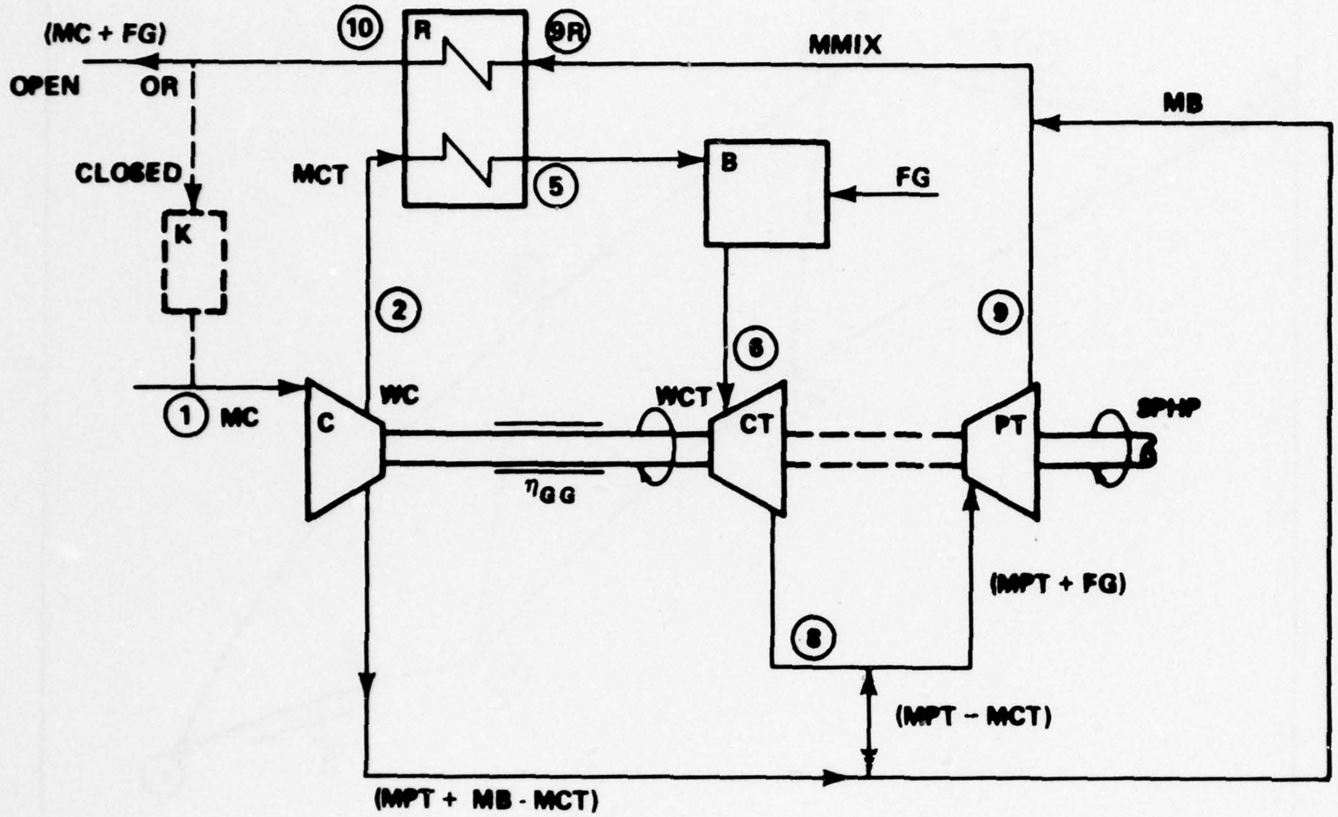

Figure 2
Universal Brayton Cycle Flow Model

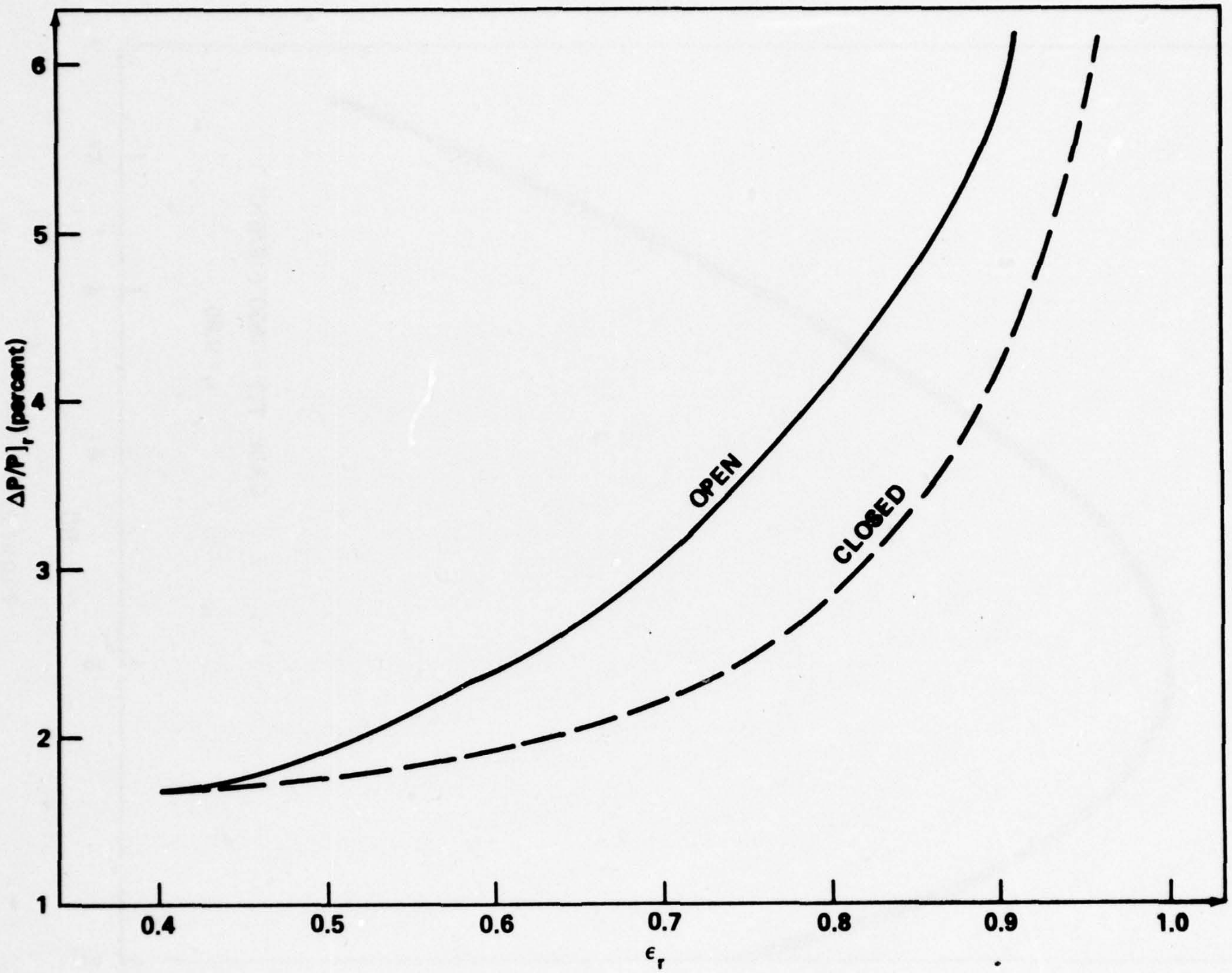

Figure 3
Recuperator Pressure Loss $\Delta P/P]_r$ vs Effectiveness ϵ_r for Open and Closed Brayton Systems

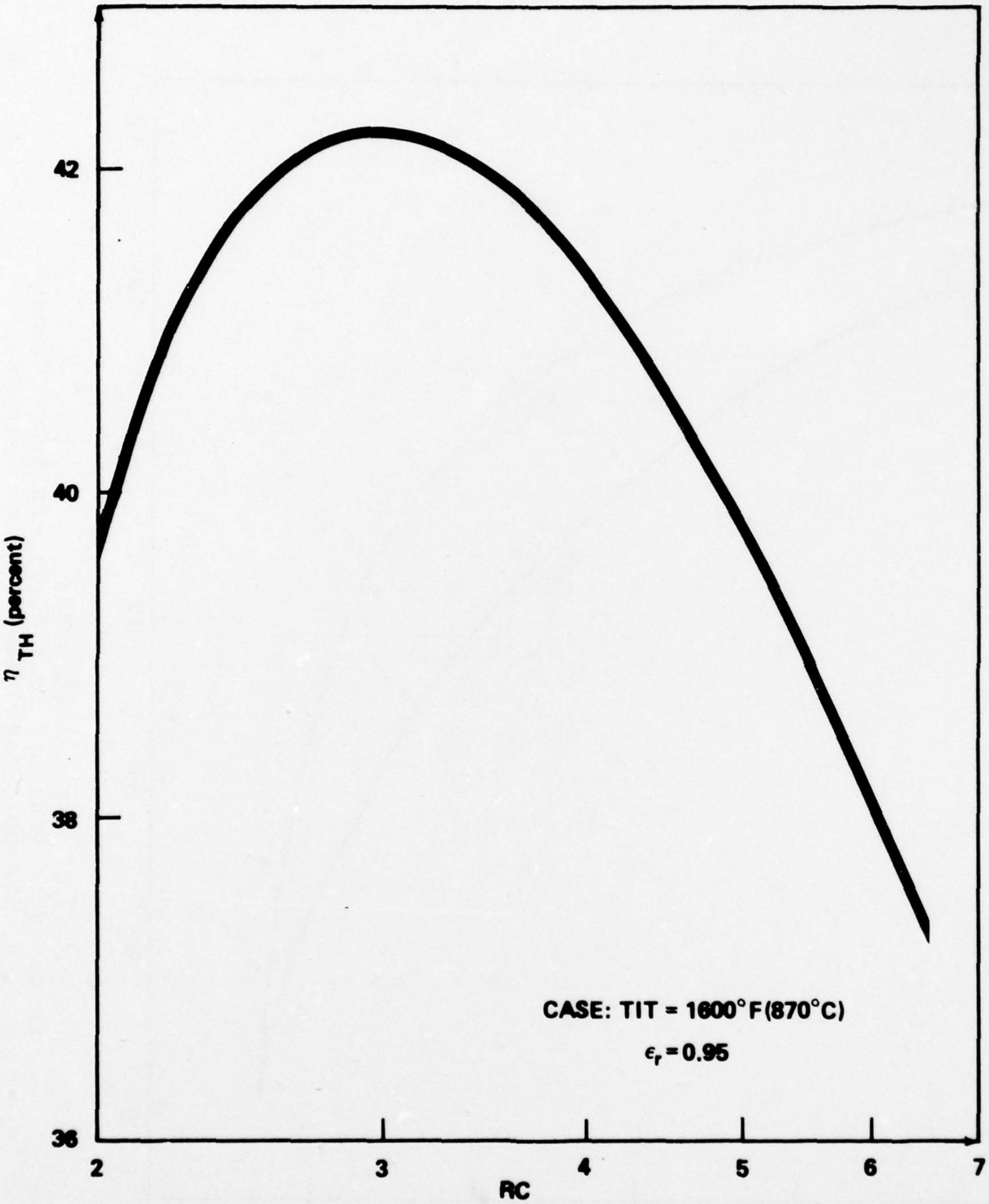

Figure 4
Example of Brayton Cycle Efficiency η_{TH} over a Possible Range of Compressor Pressure Ratio RC

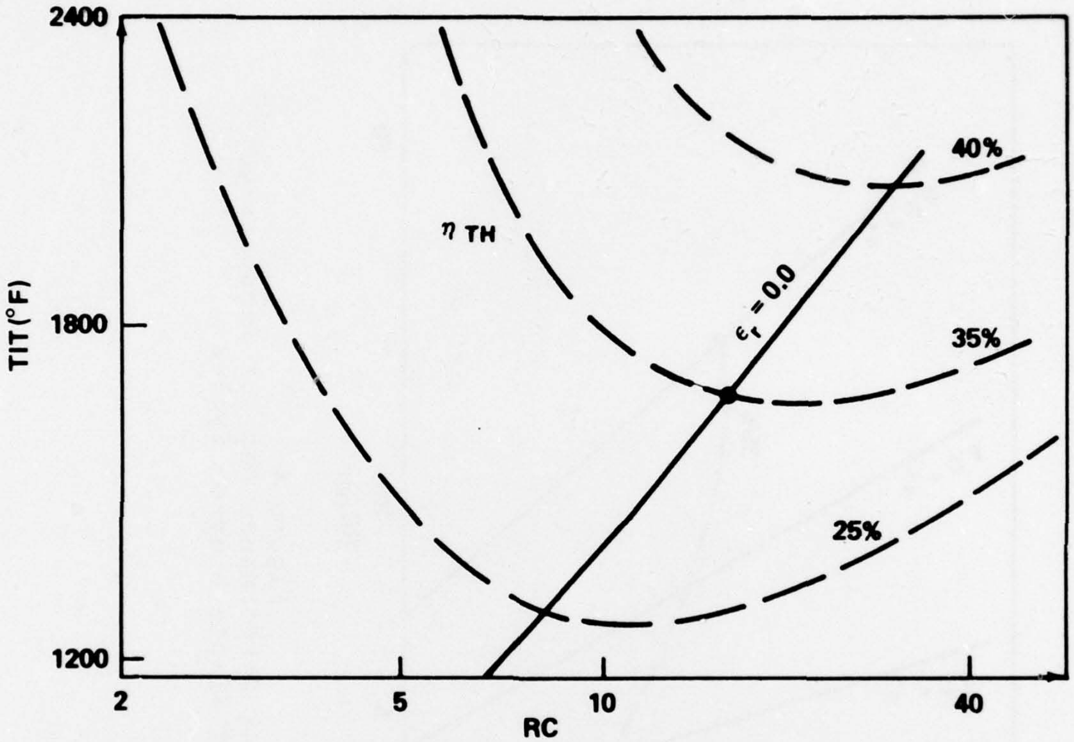

Figure 5
Thermal Efficiency η_{TH} as a Function of Turbine Inlet Temperature TIT and Compressor Pressure Ratio RC

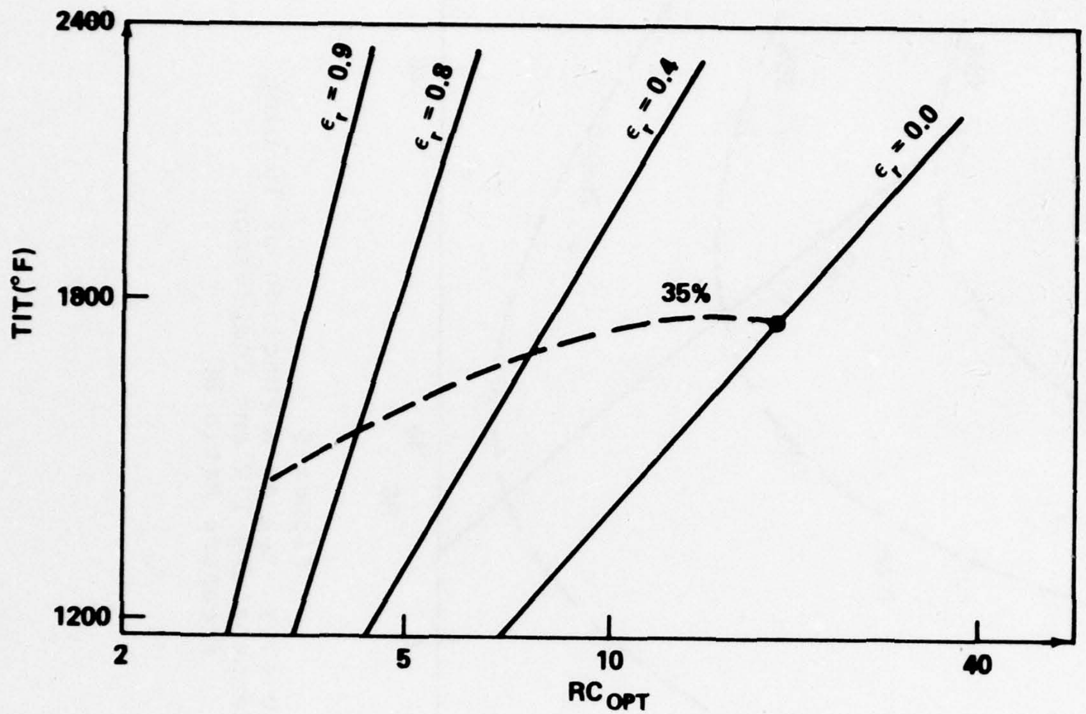

Figure 6
Lesign Point Performance Map for Recuperated and Simple Brayton Cycles

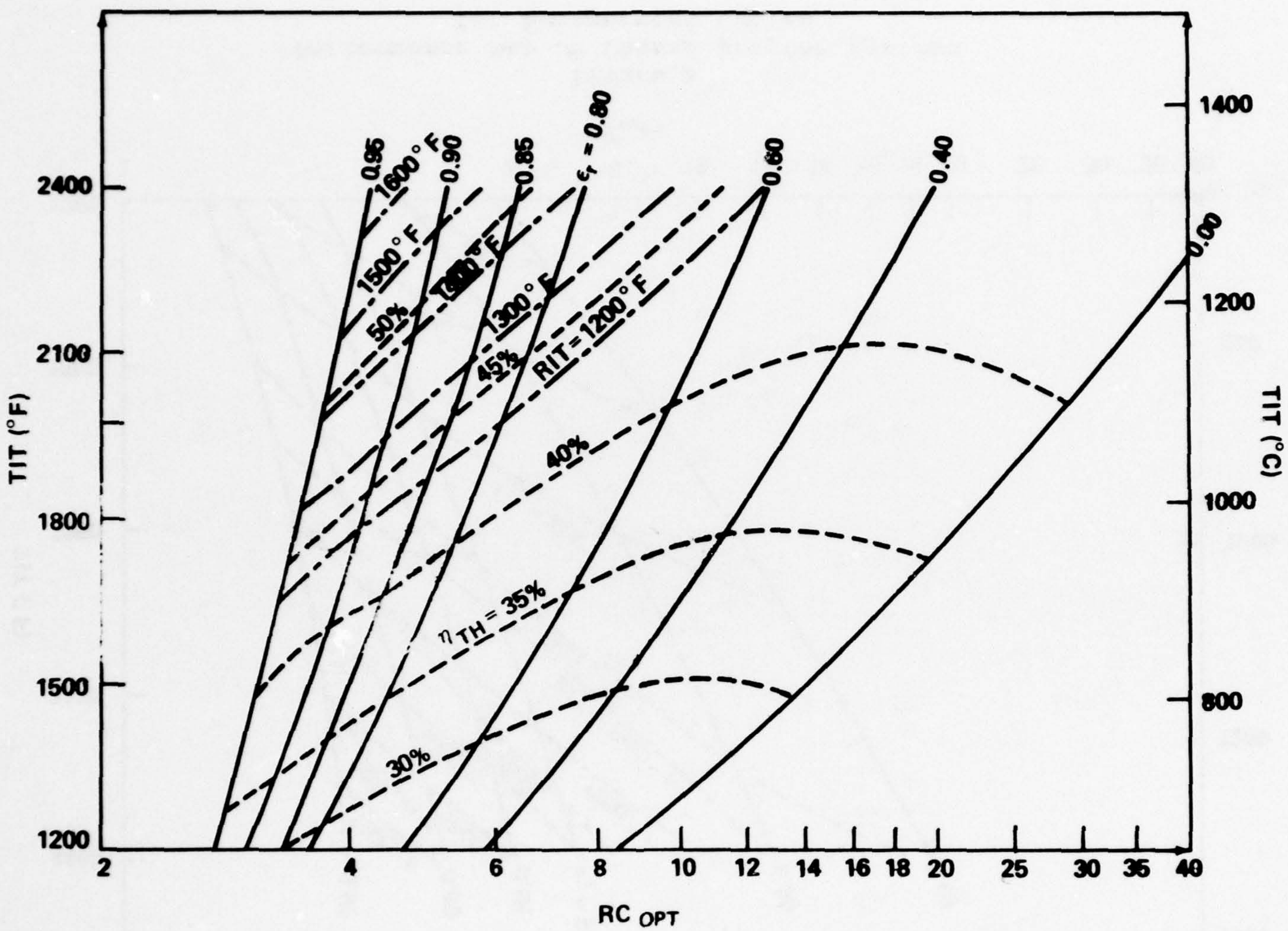

Figure 7
Performance Map of Open Brayton Systems for Recuperated and Simple Cycles

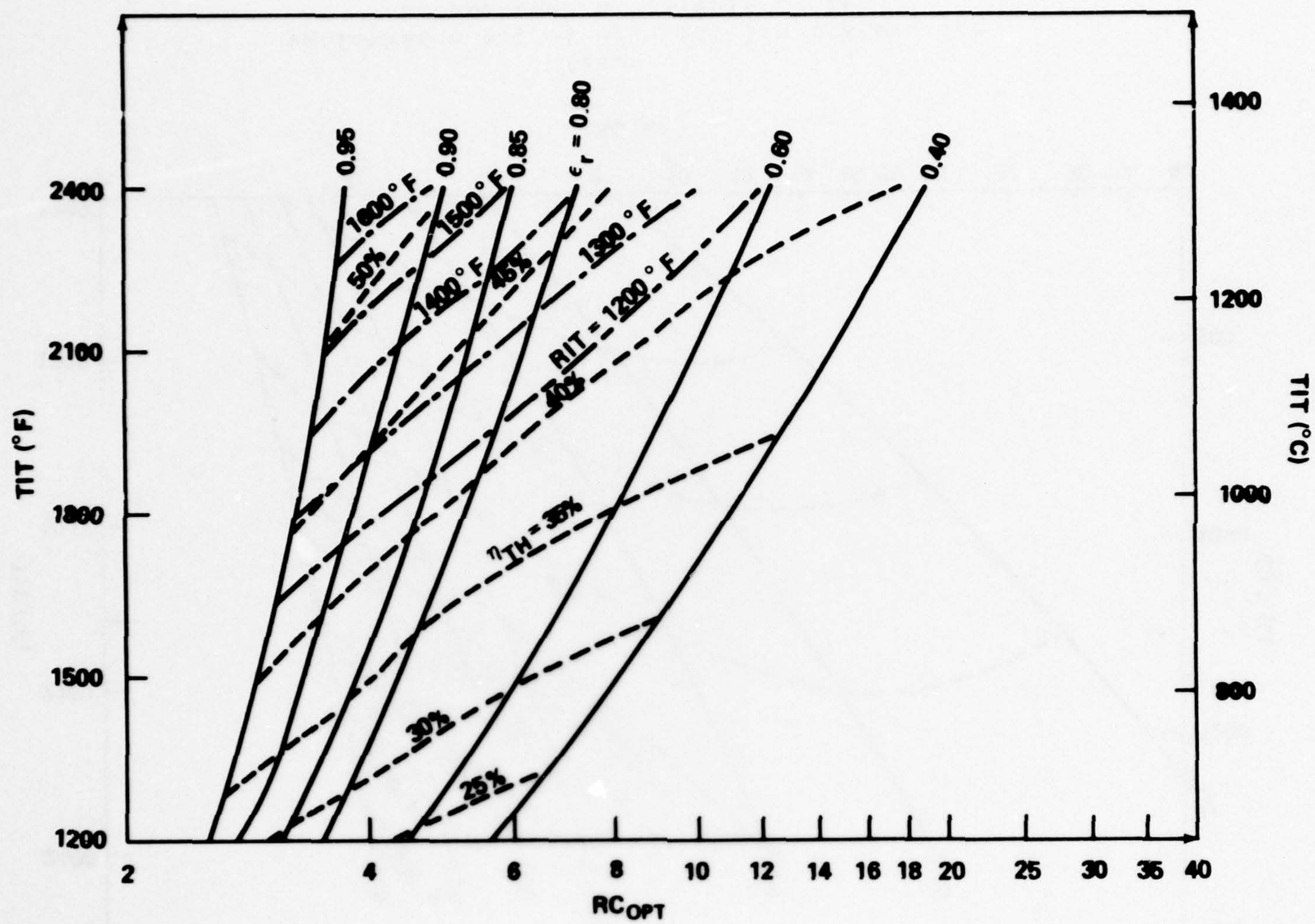

Figure 8
Performance Map of Closed Brayton Systems
for Recuperated Cycles

APPENDIX A

FORTRAN SUBROUTINE FOR CALCULATING BRAYTON CYCLE PERFORMANCES AND STATE-POINTS

```
SUBROUTINE CALCS       74/74    OPT=0 ROUND=*/ TRACE              FTN 4.5+R406         05/06/76  12.45.37

          SUBROUTINE CALCS(RC,ICASE,IERR)
          REAL MPT,MCT,MC,MB,MMIX
          COMMON /STTPTS/P2,P5,P6,P8,P9,T2,T5,T8,T8S,T9,T9R,WC,WCT,RCT,RPT
          COMMON /RZLTS/SFC,ETATH,SPHP,F
          COMMON /PRMETA/ETAC,ETACT,ETAPT,ETAGG,MCT,MPT,MC,MB,MMIX
          COMMON /PARAMC/CPC,GAMC,CPT,GAMT,P1,T1
          COMMON /PARAMP/DLPHXH,DLPHXL,EPSHX,DLPB,ETAB,T6,Q,DLPK
          FG=.01
          IF(ICASE.EQ.2)FG=0.
          IERR=0
C    CALCULATIONS      CALCULATIONS      CALCULATIONS
          T2=T1+T1*(RC**((GAMC-1.)/GAMC)-1.)/ETAC
          WC=CPC*(T2-T1)*778.*MC
          WCT=WC/ETAGG
        7 T8=T6-WCT/(MCT+FG)/CPT/778.
          T8S=T6-(T6-T8)/ETACT
          RCT=(T6/T8S)**(GAMT/(GAMT-1.))
          P2=P1*RC
          P5=P2*(1.-DLPHXH)
          P6=P5*(1.-DLPB)
          P8=P5/RCT
          P9=P1/(1.-DLPHXL)/(1.-DLPK)
          RPT=P8/P9
          IF(RPT.LT.1.)GO TO 23
          MMIX=MB+MPT+FG
          T9=T8*(1.-ETAPT*(1.-1./(RPT**((GAMT-1.)/GAMT))))
          SPHP=CPT*(T8-T9)*1.4149*(MPT+FG)
          T9R=MB*T2/MMIX+(MPT+FG)*T9/MMIX
          T5=T2+EPSHX*(T9R-T2)
          F=CPT*(T6-T5)/ETAB/Q*MCT
          IF(ICASE.EQ.2)GO TO 9
          IF(ABS((F-FG)/F)-.01)9,9,8
        8 FG=F
          GO TO 7
        9 SFC=3600.*F/SPHP
          ETATH=0.70676*SPHP/Q/F
          RETURN
       23 IERR=1
          RETURN
          END
```

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